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IMPROVING OF DIESEL COMBUSTION-POLLUTION-FUEL ECOMONY AND PERFORMANCE BY ETHANOL FUMIGATION

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Abstract: The aim of the present study is to determine experimentally the effects of ethanol fumigation (EF) on the performance parameters and NO_x emission of a naturally aspirated pre-chamber diesel engine and also to compare the EF results to that of turbocharged diesel engine. By this way, it has also been indented to determine whether or not EF gives power increment at the level of turbocharging. The engine which was originally turbocharged has been converted to naturally aspirated diesel engine by dismantling its turbocharging system. Here, ethanol was introduced into the intake air by a carburetor, which main nozzle section is adjustable to give approximately (2, 4, 6, 8, 10 and 12) % (by vol.) ratios. Experiments were performed at two diesel fuel delivery rates (FDRs), six different engine speeds for six different ethanol fumigation ratios (EFRs). Variations of performance parameters and NO_x emission for these six ethanol percentages in respect to neat diesel fuel (NDF) were determined and compared. It was determined that for 1/1 FDR, ~(7.5-11) % ethanol addition into the intake manifold reduces NO_x emission and improves engine performance for this engine. But, fuel cost for ~(7.5-11) % ethanol addition is expensive. For 3/4 FDR; ~ (7.5-12) % ethanol fumigation can be applied at (1500-3000) rpms in this engine and for these conditions NO_x emission and bsfc decrease simultaneously without any significant effective power penalty. Also, it was concluded that EF improves engine performance parameters and NO_x emission but does not increase effective power at the level of turbocharged version.
Keywords: Pre-chamber diesel engine; Ethanol fumigation; NO_x emission, Fuel economy; Combustion

ETANOL FUMİGASYONU İLE DİZEL YANMASININ, KİRLİLİĞİN, YAKIT EKONOMİSİNİN VE PERFORMANSIN İYİLEŞTİRİLMESİ

Özet: Sunulan çalışmada; indirekt püskürtmeli doğal emişli bir dizel motorunda etanol fumigasyonunun (EF'nun) motor performans parametreleri ve NO_x emisyonu üzerindeki etkileri deneysel olarak incelenmiştir. Ayrıca burada bazı veriler turboşarjlı motorun sonuçları ile karşılaştırılmıştır. Böylece EF'nun turbo etkisine yakın bir sonuç verip vermediği de belirlenmeye çalışılmıştır. Burada etanol, ana meme kesiti değiştirilebilen basit bir karbüratör ile emme kanalındaki hava içerisine gönderilmiştir ve hacimsel olarak yaklaşık % (2, 4, 6, 8, 10 ve 12) etanol oranları seçilmiştir. Deneyler; iki farklı gaz kolu konumunda, 6 farklı devir sayısında ve 6 farklı etanol oranında yapılmıştır. Seçilen etanol oranları için bulunan motor karakteristikleri ve NO_x sonuçları saf dizel yakıtı sonuçları ile karşılaştırılmıştır. Çalışmada kullanılan motor için 1/1 gaz kolu konumunda yapılan deneylerden; etanolün yaklaşık % (7.5-11) oranlarında emme kanalından geçmekte olan havanın içerisine püskürtülmesi durumunda NO_x emisyonunun azaldığı ve motor performans parametrelerinin iyileştiği belirlenmiştir. 3/4 gaz kolu konumunda ise; % (7.5-12) etanol oranlarında ve (1500-3000) d/d aralığında EF'nun uygulanabileceği ve böylece efektif güçte önemli bir kötüleşmeye neden olmadan özgül yakıt tüketiminin ve NO_x emisyonunu azaltacağı belirlenmiştir. Ayrıca EF'nun motor performans parametrelerini iyileştirdiği ve NO_x emisyonunu azaltılabileceği ancak efektif gücü turboşarj sistemi kadar arttıramadığı görülmüştür.
Anahtar Kelimeler: Ön yanma odalı dizel motoru; Etanol fumigasyonu; NO_x emisyonu, Yakıt ekonomisi;Yanma

NOMENCLATURE

be, bsfc	brake specific fuel consumption (kg/kWh)
CA, θ	crank angle (degree)
DI	direct injection
EF	ethanol fumigation
EFR	ethanol fumigation ratio
FDR	fuel delivery rate
IDI	indirect injection engine
NDF	neat diesel fuel
N_e	brake effective power (kW)
NO_x	oxides of nitrogen (ppm)
ppm	parts per million
rpm	revolution per minute
TDC	top dead center
η_e	effective efficiency

INTRODUCTION

As the number of vehicles and the world population increases; restricted fossil fuel reserves and air pollution became an ever-increasing problem. Consequently, due to limited crude oil resources and their increasing prices, considerable attention has been paid on the development of alternative fuel sources with particular emphasis on bio-fuels and new engine technologies in various countries. On the other hand, environmental pollution has reached to high levels with increasing of the number of vehicles. In addition, global warming threatens our world. Sudden rains and floods in Pakistan and China, a lot of forest fires because of the hot summer of the last year in Russia can be listed as some examples of the results of this global warming. It can be said, in the view of the above brief explanations, that environmental pollution and global warming frighten our world more than anticipated. As it is known very well, one of the most important sources of the environmental pollution and global warming is road vehicle. Consequently, recent statistics indicated that approximately 50 % of air pollution comes from road vehicles. Therefore, scientists have been studying on the new engine-related technologies; for example the use of common-rail fuel injection system, fuel injection strategies, exhaust gas recirculation, exhaust gas after-treatments, etc. and alternative fuel-related techniques. For example the use of alternative gaseous fuels of renewable nature which are friendly to the environment or oxygenated fuels which are able to reduce particulate emissions in internal combustion engines or to reduce and resolve the above-mentioned drawbacks (Heywood 1998; Merker et al. 2006; Pulkrabek 2004; Rakopoulos et al. 2010; Chen et al. 2012; Sahin et al. 2015).

Ethanol is a promising renewable oxygenated fuel for internal combustion engines and has been paid more attention in many countries. Ethanol can be produced from any of the raw materials such as sugarcane, sorghum, corn, barley, cassava, beets by means of fermentation and poses no threat to surface or ground water. Using of ethanol in internal combustion engines can reduce environmental pollution, strengthen agricultural economy, create job opportunities and reduce classical internal combustion fuel requirements (He et al. 2003). As a result, recently various studies on the using of ethanol either in spark ignition engines as well as in diesel engines have been done (Durgun 1998; Park et al. 2011; Bilgin et al. 2002; Abu-Qudais et al. 2002; Rakopoulos et al. 2007; Chauhan et al. 2011; Ekholm et al. 2008; Sahin and Durgun 2007; Jiang et al. 1990; Goldsworthy 2013, Tutak 2014). Ethanol is suitable to use by mixing (blending) with gasoline in spark ignition engine because its high octane number. In spite of this, ethanol cannot be used by direct mixing with DF in compression ignition engine due to the various difficulties encountered. The main difficulties are: (1) Ethanol has extremely low cetane number, diesel engine is known to prefer high cetane number fuels (45-55) which auto-ignite easily and give small ignition delay (Abu-Qudais et al. 2002). (2) Adding ethanol to DF can reduce the lubricity of the fuel and create potential wear problems in sensitive fuel pump designs (Rakopoulos et al. 2007; Rakopoulos et al. 2010; Abu-Qudais et al. 2002). (3) Ethanol possesses lower viscosity and calorific value, with the latter imposing some changes on the fuel delivery system for keeping the engine maximum power and also more amount of ethanol than DF is required by mass and volume. (4) Ethanol at large percentages cannot mix with DF homogenously. Hence using of diesel fuel-ethanol blends is not feasible. Also, the blends were not stable and separate in the presence of trace amounts of water (Abu-Qudais et al. 2002).

For these reasons, in the using of ethanol in diesel engines, various techniques have been developed to make diesel engine technology compatible with the problems of ethanol-based fuels. In general these techniques can be classified into three categories. (1) Ethanol–diesel fuel blends: Mixing fuels in the fuel tank, displacing up to 25% of DF demand. (2) Dual injection: Using separate injection systems for each fuel, displacing up to 90% of DF demand. (3) Ethanol fumigation (EF): Adding ethanol in to the intake air charge, displacing up to 50 % of DF demand (Abu-Qudais et al. 2002; Rakopoulos et al. 2007; Sahin and Durgun 2007).

The most attractive and the simplest of these techniques is EF for diesel engines. Fumigation is a technique by which ethanol is introduced into intake air flow by a simple carburetor or injecting ethanol into intake air stream. This technique requires using a carburetor, a vaporizer or an injector, along with a separate fuel tank, lines and control. However, using a simple carburetor requires minor modifications on the engine intake system, and thus, this method is fairly cheap (Sahin and Durgun 2007; Sahin et al. 2015).

Many studies on EF and fumigation of other fuels such as methanol, gasoline, diesel fuel etc., in direct injection (DI) diesel engines can be found in the literature (Odaka et al. 1992; Zhang et al. 2011; Sahin et al. 2008; Chapman et al. 2008). In these fumigation studies, generally promising results have been obtained in terms of engine performance characteristics and exhaust emissions. It can be seen from the literature that, fumigation technique decreases substantially NO_x and soot emissions by using above-mentioned fuels (Abu-Qudais et al. 2002; Chauhan et al. 2011; Sahin and Durgun 2007; Zhang et al. 2011; Jiang et al. 1990; Tutak 2014). Abu-Qudais et al. (2002) showed that 20 % percentage of EF gave increases of 7.5% in brake thermal efficiency, 55% in CO and 36% in HC emission and reduction of 51% in soot mass emission. Chauhan et al. (2011) found that 15 % percentage of EF exhibited better engine performance with lower NO_x, CO, CO_2 and exhaust temperature. But EF increased unburned hydrocarbon (HC) emission in the entire load range. Sahin and Durgun (2007) numerically studied the effects of 2.5–20 % EF on a DI diesel engine cycle and performance characteristics. Here the effects of 2.5–20% EF were investigated numerically. Their results showed that by increasing EF, effective power, effective efficiency and carbon monoxide (CO) increased and specific fuel consumption (SFC) and nitric oxide (NO) emission decreased. Also gasoline, methanol and dimetilether have been used by applying fumigation technique in DI diesel engines. In these studies fumigation technique gives better results in the point of view of engine performance and exhaust emissions. By this way NO_x and smoke emissions decrease dramatically (Chauhan et al. 2011; Sahin and Durgun 2007; Zhang et al. 2011; Chapman and Boehman 2008; Sahin et al. 2015).

From the above literature survey it is clear that many studies on the effects of fumigation of various light fuels in DI diesel engines have been performed (Abu-Qudais et al. 2002; Sahin and Durgun 2007; Zhang et al. 2011). But in spite of this there are few parametric and experimental alternative fuel studies in indirect injection (IDI) automotive diesel engines (Turkcan and Canakcı 2008; Leevijit and Prateepchaikul 2010; Selim et al. 2013). Actually, IDI diesel engines are not used commonly nowadays. Because of their higher speeds, DI diesel engines with common-rail fuel injection system have been used widespreadly. Instead of this, by obtaining any probable enhancement in IDI diesel engine for EF, their usage could be increased and by this way using of cheaper engine with lesser injection pressure could be realized. For these reasons, in the present study, the effects of EF on an IDI naturally aspirated automotive diesel engine performance and NO_x emission were investigated experimentally and also the obtained EF results for naturally aspirated version were compared to that of turbocharged version. By this way, it was also aimed to determine if EF could results in similar effects of turbocharging. In addition cost analysis was conducted and EF results were compared with NDF.

EXPERIMENTAL SYSTEM AND PROCEDURE

Engine and Experimental Set Up

EF experiments were carried out on a 4 cylinder, 4-stoke, water cooled, pre-chamber FORD automotive diesel engine (model XLD 418 T; Ford). Main technical specifications of the engine are given in Table 1. This engine was originally turbocharged. *In the presented study, it is aimed to investigate the effects of EF on a naturally aspirated diesel engine.* For this reason the turbocharging system which consists of a turbine and a compressor were disassembled. Effective power at nominal engine speed (4000 rpm) was determined experimentally as 35.616 kW for naturally aspirated engine version. However effective power of the turbocharged engine at nominal speed is 55 kW. Thus, EF experiments were performed on turbocharged version of this engine. The EF results for turbocharged engine version were presented in references (Sahin et al. 2010, Sahin et al. 2015). The other goal of this study was to show either or not EF would give enhancement near to turbocharging. By this way it is thought to enhance the engine performance and also NO_x pollution. As a practical and cheap solution it is planned to apply EF by installing an elementary carburetor to engine intake system.

Test system used in the experiments was produced by Cussons and its technical drawing was presented in Figure 1. Here; loading of the engine was done by a water brake and the brake moment (loading force) was determined electronically. The fuel consumption was measured by mass.

Nitrogen oxide (NO_x) concentration and oxygen (O_2) percentage in the exhaust gases were determined by gas analyzer (MEXA-720 NO_x, Horiba). It is a direct-installation gas analyzer using a zirconia ceramic sensor and it can measure NO_x concentration and fuel air ratio in the exhaust gases. The main specifications of this gas analyzer are given in Table 2.

Cylinder gas pressure was measured by using of a water-cooled pressure sensor (8QP500C type quartz, AVL). The pressure sensor was mounted on the head of the first cylinder of the engine by removing of the hot plug. The signal outputs of the pressure sensor were amplified by electronic indicating system (P4411 type, Cussons). This system consists of a processing rack and an oscilloscope and it incorporates microprocessor controlled circuitry to monitor the angular position of the crankshaft to provide an accurate and versatile measurement of either time or angle against which the cylinder gas pressure and DF injection parameters can be analyzed using a variety of display or data acquisition equipment. Here a computer and data acquisition card (NI PCI-6221 type, National Instruments) which has 16-bit resolution and 250 ksample/s sampling were used in order to convert signals from analog to digital and to record the obtained data. Charge amplifier output of electronic indicating system which is proportional to cylinder gas pressure and angles determined by using top dead center (TDC) signal taken from a magnetic pick-up were stored for one cycle with a sampling time 25 μs. Cylinder gas pressure was acquired using pressure transducer at averagely 0.5 degree crank angle (CA) resolution. In each test, 5 consecutive cycles were collected and averaged. Moving average filtering method was applied to the measured cylinder gas pressure data to reduce noise effects.

Ethanol Adding System

In the present study, an elementary carburetor was mounted on the inlet manifold. Some modifications have been done on this carburetor for introducing at desired amount of ethanol into inlet air. The air and gas throttles of the carburetor were dismantled and the other auxiliary equipments of the carburetor were left out of order. The carburetor air inlet was connected to the air consumption measuring box by a flexible hose. EFRs were varied by a fine threaded adjustment screw which can vary the main jet section. This screw was designed by authors and it was manufactured in Trabzon industry. Technical drawing of the used carburetor in the experiments was presented in Figure 1. At the EF tests, 6 different carburetor main jet openings were chosen to obtain 6 different EFRs of ~ (2, 4, 6, 8, 10, 12 %, by vol.).

Operating Conditions

In the present study, experiments were done for two FDRs such as full (1/1) and three quarter (3/4) FDRs and at 6 different engine speeds (1500, 2000, 2500, 3000, 3500 and 4000 rpms). Experiments were firstly carried out for neat diesel fuel (NDF) to obtain a database for comparison with those obtained for each EFRs. Commercially available DF was used in the experiments. Then, the adapted carburetor, which introduced above, was mounted to intake manifold of the engine. Also, as shown in Figure 1, a small ethanol tank, a scaled glass bulb and a flexible pipe were used to introduce ethanol into intake air. Any other change on the engine and experimental system was not done and the engine mainly operates due to diesel principle. The applied experimental procedure is given briefly in the following paragraphs.

Experiments were carried out after running the engine for warming during approximately 30 minutes until it reaches steady state and cooling water temperature becomes (70 ±5) ^oC. Thus, the performance, emission characteristics and in-cylinder gas pressure values were measured for EF and the results were compared with NDF. For example; for 1/1 FDR as engine running at 3000 rpm, firstly carburetor mean jet opening was adjusted to 1st opening and it was fixed. This opening gives approximately 2 % EFR. Then, the engine speed was reduced to 1500 rpm and tests for approximately 2 % EFR were performed from 1500 rpm up to 4000 rpm at 500 rpm steps by adjusting the engine load suitably. Thus, tests for 2% EFR at 1/1 FDR carried out at six different

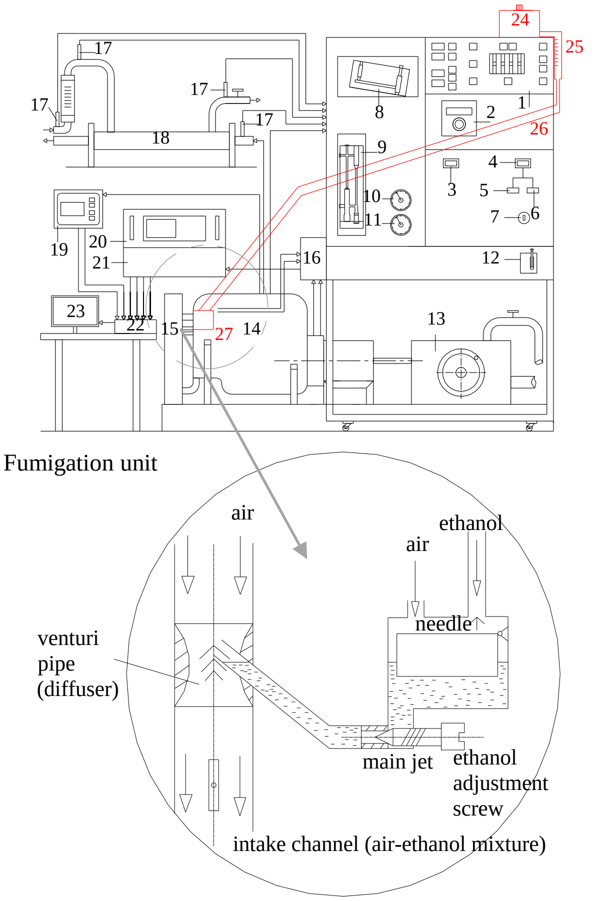


Figure 1. Schematic view of the test system. 1-fuel measurement unit, 2-digital display for temperatures, 3-speed, 4-force, (5,6)-loading unit, 7-start switch, 8-inclined manometer, 9-coolant flow meter, 10-oil temperature, 11-inlet manifold pressure, 12-gas throttle, 13-hydraulic dynamometer, 14-engine, 15-cooling package,16-inference unit for gas pressure, fuel line pressure and crank angle pick-up sensors, 17-thermocouples, 18-exhaust gas calorimeter, 19-gas analyzer, 20-ossiloscope, 21-electronic indicating system, 22-data acquisition card, 23 computer, 24-ethanol tank, 25-scaled glass bulb, 26- flexible hose, 27-carburator.

Table 1. Specifications of the test engine.

Engine	Ford XLD 418 T turbocharged IDI automotive diesel engine
Displacement	1.753 liter
Number of cylinder	4
Bore & stroke	82 & 82.5 mm
Compression ratio	21.5: 1
Injector opening pressure	150 bar
Maximum power	55 kW @ 4500 rpm
Maximum torque	152 N m @ 2200 rpm
c.r.lenght	130 mm

Table 2. Target components, ranges and accuracy of measurement of the MEGA-720 NO_x gas analyzer

Nitrogen oxides (NO_x)	(0-3000) ppm
Accuracy of NO_x;	± (3-5) % ppm
Air-fuel ratio (A/F)	(9.5 - 2000)
Accuracy of A/F;	± (0.15-0.4) A/F
Oxygen (O_2)	(0-25) vol %
Accuracy of O_2;	± 0.5% O_2

engine speeds. After that, firstly engine speeds were adjusted to 3000 rpm and then, 2nd carburetor means jet opening was selected and it was retained fixed. This opening gives approximately 4 % EFR. Then, the engine speed was reduced to1500 rpm and tests for approximately 4 % EFR were performed from 1500 rpm to 4000 rpm by 500 rpm increments. Tests for EFR of 6%, 8%, 10% and 12% carried out by applying the same procedure and repeating similar adjustments. After these, experiments for 3/4 FDR were similarly repeated. Thus, the performance, NO_x emission characteristics and cylinder gas pressure values were measured for EFR and the results were compared with NDF.

For any main adjustment position as engine speed was changed, in spite of remaining the DF amount unchanged, as a result of the principle of elementary carburetor, the amount of the ethanol and ethanol ratio to DF varied. That is; for the 1st opening, in spite of being fumigation ratio 1.5 % at low engine speeds, it becomes 2.5% at higher speeds. This peculiarity can be clearly seen in Table 3-4. For this reason by adjusting the carburetor main jet opening, it is tried to get 2 % fumigation ratio at 3000 rpm. Main jet opening was adjusted suitably to obtained desired fumigation ratio at 3000 rpm (at the middle engine speed) and this adjustment was used at all of the other engine speeds. ***As shown in Tables 3-4, 1st, 2nd, 3rd, 4th, 5th, 6th carburetor main jet openings give ~ (2, 4, 6, 8, 10,12 % by vol.) ethanol fumigation ratios respectively.*** In these tables and in the following figures for example the abbreviation 1st EFO shows EF at 1st opening.

In the present study, carburetor main jet openings were adjusted such as EFRs would not exceed the upper value of 20 % because at higher fumigation ratios diesel knocking problem would be occurred. However, in the present study any engine knocking event has not been observed for selected ethanol percentages during experiments. In addition, for applying error analysis to the experimental results every measurement was taken 3 times. Thus, for 2 FDRs, 6 engine

Table 3. EFRs at 1/1 FDR for different carburetor main jet openings.

n[rpm]	1. EFO [%]	2. EFO [%]	3. EFO [%]	4. EFO [%]	5. EFO [%]	6. EFO [%]
1500	1.442	4.070	4.612	7.394	10.650	12.210
2000	1.899	5.223	4.409	7.665	8.004	9.768
2500	1.967	4.002	5.155	6.987	10.514	11.193
3000	1.984	3.765	5.696	8.032	10.242	11.532
3500	2.035	4.206	5.495	8.344	10.107	11.260
4000	2.442	4.070	5.902	8.004	10.311	10.989

Table 4. EFRs at 3/4 FDR for different carburetor mean jet openings.

n[rpm]	1. EFO [%]	2. EFO [%]	3. EFO [%]	4. EFO [%]	5. EFO [%]	6. EFO [%]
1500	2.467	4.748	6.461	8.072	10.887	14.313
2000	2.917	5.630	7.055	9.022	12.074	16.551
2500	3.731	6.831	8.945	11.464	14.584	20.553
3000	2.035	4.273	5.223	7.123	7.869	13.974
3500	1.832	3.595	4.477	5.359	6.716	10.379
4000	1.289	2.781	4.409	5.495	7.326	10.786

speeds, 6 EFOs totally 216 EF tests were done. Also, for NDF at two FDRs (1/1 FDR and 3/4 FDR) and 6 engine speeds 36 tests were performed. Thus, being for EF and NDF, totally 252 tests were realized.

Calculation of Engine Characteristics

For evaluation of the experimental results for NDF and EF the calculation method, which details given by Durgun (Durgun and Ayvaz 1996; Durgun 1990), was used. Here only the principles of the calculation procedure are summarized. Effective power output from the engine crankshaft being converted to the standard conditions and corrected for ambient air humidity was calculated by using the following relation

$$N_e(kW) = 0{,}1013 \frac{T_b \omega}{p_0} \sqrt{T_0/293} X_{hum} \quad (1)$$

where ω is angular velocity of the crankshaft, T_b (Nm) is brake torque, T_0 (K) and p_0 (MPa) are ambient air temperature and pressure, respectively. X_{hum} is the humidity correction factor and it is determined depending on dry and wet thermometer temperatures. Here, fuel consumption of the engine was determined by mass and consumption duration of 40 g of DF was measured. The amount of the ethanol used during this time interval was determined by using a scaled glass bulb. By this way, brake specific fuel consumption (bsfc) was calculated as follows:

$$b_e(kg/kWh) = \frac{(m_d + m_e)3600}{1000\Delta t N_e} = \frac{(40 + V_e\rho_e)}{1000\Delta\, tN_e} \quad (2)$$

Here, m_d and m_e are the masses of consumed DF end ethanol during Δt (s) respectively, Δt (s) is the duration of consumption of 40 g of DF, V_e is the volume of ethanol used during Δt (s) and ρ_e is density of ethanol. Variation ratios of brake specific fuel consumption (bsfc) and other engine characteristics were calculated in the similar way, for example, as follows:

$$\frac{\Delta b_e}{b_e} 100[\%] = ((b_{e,fum} - b_{e,d}) / b_{e,d})100 \quad (3)$$

where $b_{e,fum}$ and $b_{e,d}$ are bsfc for EF and DF respectively.

Cost Analysis

In presented study, a practical cost analysis was performed by using the following relationship, which was developed originally by Durgun and Ayvaz (1996). In this relationship cost evaluation was done according to the price of DF.

$$\frac{\Delta C}{C_1} 100[\%] = \frac{C_2 - C_1}{C_1} 100 = \left[\frac{x_1 + \sum\limits_2^n X_i r_i}{x_1 + \sum\limits_2^n X_i s_i} \left(1 + \frac{\Delta b_e}{b_e} \right) - 1 \right] 100 \quad (4)$$

where

$$r_i = C_i/C_1, \quad r_1 = C_1/C_1 = 1,$$
$$r_2 = C_2/C_1 = 30/4.22 = 7.109$$

$$S_i = \rho_i/\rho_d, \quad S_1 = \rho_d/\rho_d = 1,$$
$$S_2 = \rho_e/\rho_d = 799.5/814 = 0.982$$

C_1 is cost of DF, C_2 is cost of ethanol and $\Delta b_e/b_e$ is difference ratio of bsfc, ρ_e and ρ_d are densities of ethanol and DF respectively. Here, units of (C_1, C_2), (ρ_e, ρ_d) and b_e are (TL/lt), (kg/m^3) and (kg/kWh) respectively. The costs and other principle characteristics of DF and ethanol are given in Table 5.

Error Analysis and Uncertainties

In the present study each experiment value was measured 3 times. By applying Kline and Mc.Clintock's method, given by Holman (2001), error analysis was applied to the measured values and uncertainties were determined. Here, as being measured each value 3 times, Student's t-distribution must be applied to the experimental data.
Errors in various terms were determined by applying well known method of evaluation of experimental data. For example, uncertainty interval of torque values is determined as (0.5-5) %. Error analysis for derived values such as effective power, bsfc and effective efficiency was also

performed. At the end of the error analysis it was determined that for example the uncertainty in effective power values is in the interval of (0.2-1.2) %. By examining all of the other error analysis results, it was seen that probable errors of the measured main values and also uncertainties in the bsfc and effective efficiency were in the interval of (0.02-5) %. From these results, it can be said that the errors in the measuring of the principle values and the probably uncertainties in the derived values would not affect significantly the uncertainties of the results.

RESULTS AND DISCUSSION

In this section, experimental results about performance characteristics and NO_x emission of an automotive diesel engine with ethanol injection into intake air were investigated and compared with NDF. Obtained results are presented in the following figures various ways. Also, in-cylinder pressure-crank angle (p-θ) and in-cylinder pressure-volume (p-V) diagrams, originally called indicator diagrams, were presented for 1/1 and 3/4 FDRs at 3000 rpms. The indicator diagrams for the other engine speeds are also available (Durgun et al. 2009). But they were not presented in this paper because of page restriction. In the following paragraphs; firstly, some information about the fumigation method and its probable effects on the mixing process, engine combustion and performance will be presented. After that the effects of EF on the engine performance characteristics, combustion characteristics, NO_x emission and fuel economy will be given and discussed sophisticatedly. Finally also turbocharged effect of EF was given briefly. It is useful to be repeat again the following explanation. *In the experimental study, 6 different carburetor main openings were used to obtain 6 different EFRs. For these reason, in the evaluations of results we prefer to state ethanol fumigation openings. However; to understand the results clearly, ethanol percentages relate to these openings were given in parenthesis.* In the presented study, engine speeds of 1500, 2000 and 2500 rpms are referred as low engine speeds and engine speeds of 3000, 3500 and 4000 rpms are referred as high engine speeds.

Table 5. Properties of diesel fuel and ethanol.

Properties	Diesel fuel	Ethanol
Chemical formula	$C_{14.342}H_{24.75}$	C_2H_5OH
Molecular mass [kg/kmol]	197.21	46.07
Density [kg/m^3]	814*	799.5*
Lower heating value [kJ/kg] (calculated from Mendeleyev formula)	42685.7**	27423.24**
Cost (TL***/lt) 2014, Trabzon, Turkey	4.22	30
Composition, mass [%]	$c' = 0.873$, $h' = 0.127$	$c' = 0.521$, $h' = 0.131$, $o'_y = 0.347$

*measured in laboratory, **calculated from Mendeleyev formula, *** TL: Turkish Lira

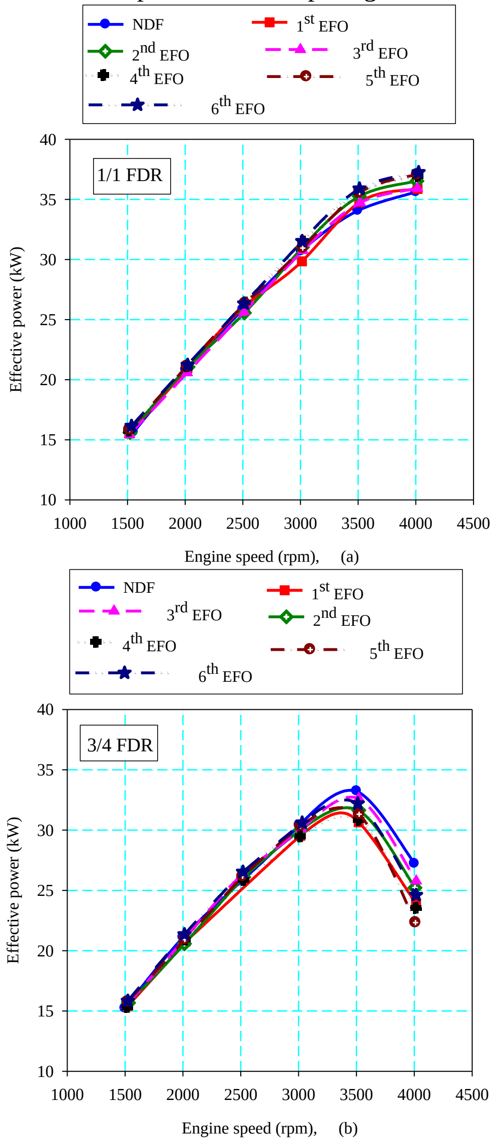
EF Process and Its Probable Effects on the Process of Air-Fuel Mixing and Combustion

In the fumigation method, light fuel (in the present study light fuel is ethanol) is injected into intake air by a carburetor which adapted as explained above or by a suitably programmed electronic fuel injection system. Thus prepared air-light fuel mixture is compressed during compression process and its temperature is raised. By this way, light fuel (here ethanol) evaporated and it gets to ready to burn through the end of compression. But ethanol or any light fuel such gasoline is not suitable to ignite by self and for this reason it could not burn. Through to the end of compression, at injection advance, DF is injected into this mixture. At the end of the ignition delay, DF accumulated during this time self ignites and burns instantaneously. Until this time light fuel has evaporated fully and it necessitates a spark for burning. Instantaneous burning of DF acts as if a spark for ignition of this air-light fuel mixture which is ready to burn. Thus, it is thought that at the end of ignition delay, by the effect of self ignition of DF fast combustion of air-light fuel mixture causes additional gas motions. This additional gas motions enhance the mixing process of DF injected, after this moment, with air more fastly and homogenously (Sahin and Durgun 2007; Sahin et al. 2008; Sahin et al. 2015). *It is well known that combustion process in a diesel engine is controlled by fast and homogenous mixing DF with air* (Heywood 1998). On the other hand, combustion process is the most important event which affects engine performance characteristics. Improving fuel-air mixing by this additional gas motions means that combustion process would get better and engine performance could be improved and exhaust pollution would be reduced (Heywood 1998, Sahin and Durgun 2007).

Effects of EF on Engine Performance Characteristics

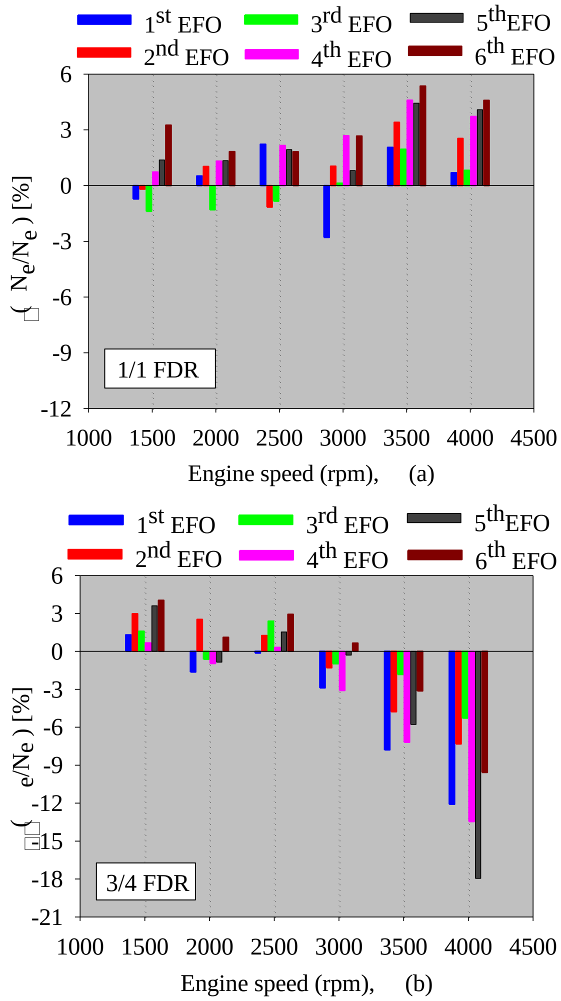
(a) Effective power: Figure 2a and Figure 3a show the variations and variations of the variation ratios of the effective power with engine speed at six different EFRs for 1/1 FDR. As shown in these figures; effective power decreases at low engine speeds but it increases at high engine speeds for 1st, 2nd and 3rd EFOs (~2-5% EFRs). However, effective power increases at all of the engine speeds for 4th, 5th and 6th EFOs (~7.74 -11.16% EFRs) and on average 2.71% increment in effective power is obtained for these openings. In Figure 3a, it can be also seen that at higher engine speeds increment ratios of effective power are higher than that of lower engine speeds for 4th, 5th and 6th EFOs (~7.74 -11.16% EFRs). The increase in effective power output for EF is occurred, because EF influences combustion process. As stated above, DF and air mixing would be improved by additional gas motions which occur by instantaneous burning of ethanol-air mixtures. Thus, combustion process would get better and engine performance characteristics such as effective power, bsfc and effective efficiency could be improved. In addition, ethanol that is introduced into the intake manifold will vaporize partly and would cool the air during intake stroke. This could increase the volumetric efficiency and could lead to an increase in the ignition delay (Abu-Qudais et al. 2002; Sahin et al. 2010). By the effects of the ignition delay increment, more DF would be physically prepared by evaporation and mixing for chemical reactions, which increases the amount of DF burned and the rate of heat release in the premixed burning period. This results in enhancing of combustion and improving of combustion efficiency (Abu-Zaid 2004).

The effects of EF on the effective power are shown in Figure 2b and Figure 3b for 3/4 FDR. As can be seen in Figure 2b, there is little change or even a slight increase in the effective power at 1500 and 2500 rpm for all of the openings. However, the variation ratios of effective power are not consistent at 2000 rpm for selected openings.



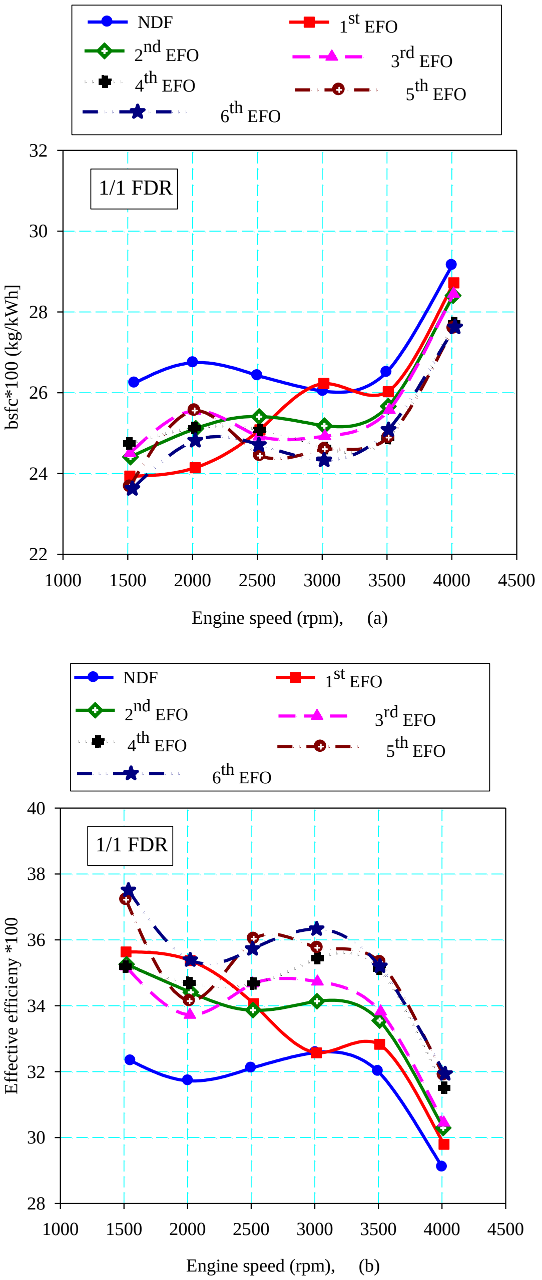
Figures 2. (a and b) Variations of effective power versus to engine speed at 1/1 and 3/4 FDRs for different EFOs (EFRs) respectively.

At higher engine speeds, effective power decreases considerably for selected ethanol percentages. On average 8 % reduction in effective power is obtained at 3500 and 4000 rpm for all of the openings. For 3/4 FDR, excess air coefficient values are very high and also, it increase with EF. For example; excess air coefficient values at 3500 rpm are 1.526, 1.634, 1.651, 1.657, 1.738, 1.765, and 1.759 for NDF and EFOs between of 1st-6th (~ 2-10% EFRs). Thus, ethanol-air mixture might to be too lean to support combustion phase, resulting in decrease of effective power. Also, it is thought that lean ethanol-air mixture could not burn instantaneously or could burn at very lean conditions and thus additional gas motions, which enhance the mixing process of injected DF and air, could not occur or insufficient gas motions could take place. Therefore, the useful effect of EF cannot be reflect in effective power for 3/4 FDR. However, low combustion temperature may be occurred due to lean combustion. Thus, NO_x emission decreases significantly at 3500 and 4000 rpm. This result will be discussed in the later paragraph.



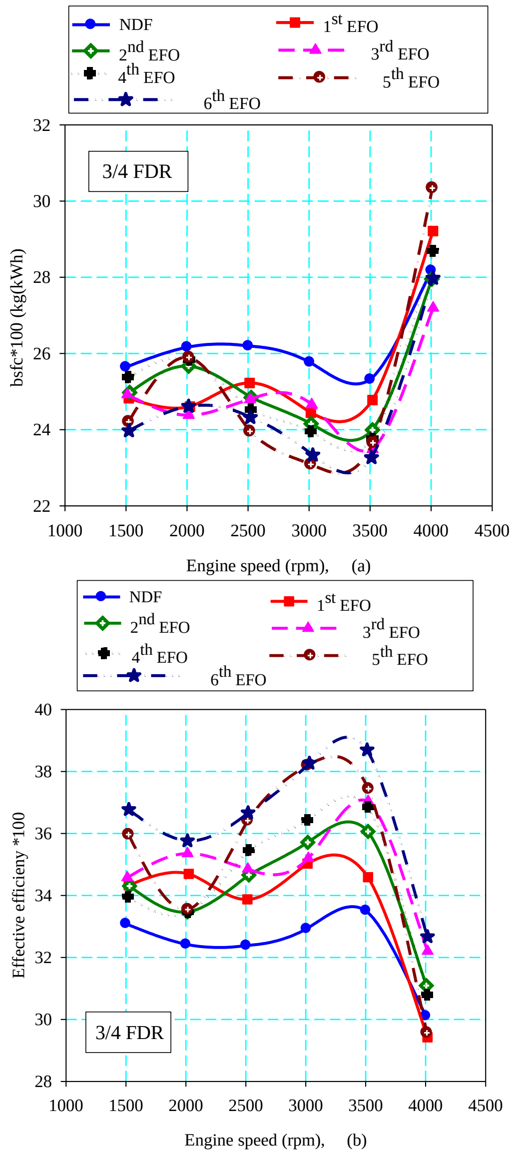
Figures 3. (a and b) Variations of the variations ratios of effective power versus to engine speed at 1/1 and 3/4 FDRs for different EFOs (EFRs) respectively.

(b) Brake specific fuel consumption and brake effective efficiency: Figures 4 (a and b) and Figures 5 (a and b) show the variation of bsfc and effective efficiency with engine speed at six different EFRs for 1/1 and 3/4 FDRs respectively. Also, Figures 6 (a and b) demonstrate the variation ratios of bsfc with engine speeds at six different EFRs for 1/1 and 3/4 FDRs respectively. It can be seen from Figures 4 (a and b) that bsfc decreases and effective efficiency increases with EF for 1/1 FDR for all of the openings and engine speeds. On average 5.3 % reduction in bsfc is obtained for all operating conditions. As can be seen in 6a, the reduction ratios of bsfc are not consistent at lower engine speeds. However, at high engine speeds, bsfc decreases with increasing ethanol percentages. Similar variations for effective efficiency have also been observed for 1/1 FDR. On average 9.48 % increment ratio of effective efficiency is obtained at 5th and 6th EFOs (~ 9.97 and 11.16 % EFRs) for this FDR. Again, as explained previously this is due to the enhanced combustion phase and also better DF and air mixture formation with EF.

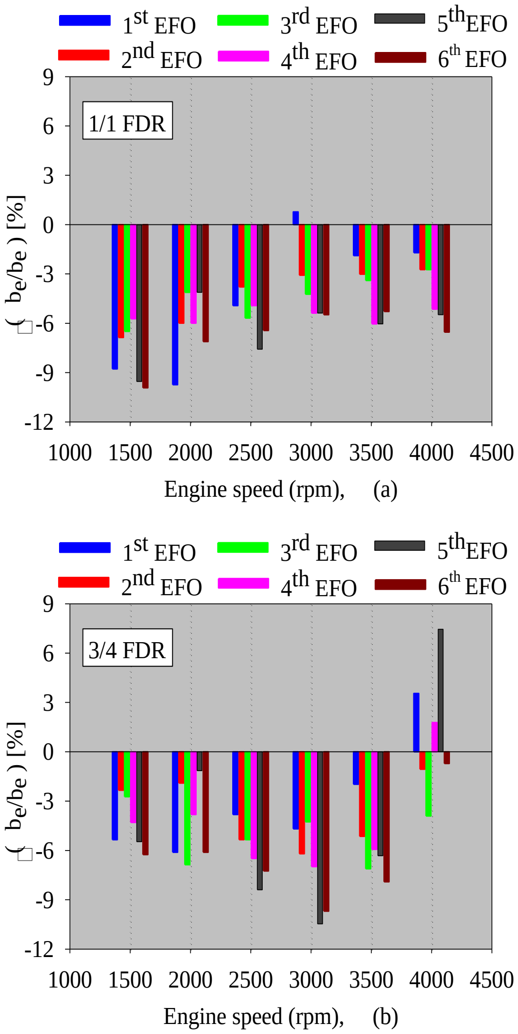


Figures 4. (a and b) Variations of bsfc and effective efficiency versus to engine speed at 1/1 FDR for different EFOs (EFRs).

The variations of the bsfc with engine speed at six different EFOs for the 3/4 FDR are shown in Figure 5a and the variation ratios of bsfc with EFRs are given in Figure 6b. As can be seen in Figure 6b that bsfc decreases for all of the EFOs and engine speeds. However, bsfc increases at only 4000 rpm for 1st, 4th and 5th openings (~2.4%, ~7.7% and ~10% EFRs) and reduction ratios are not consistent at 1500 and 2000 rpms. It can also be seen from Figure 6b that on average 4.4 % reduction in bsfc is obtained at all of the openings and engine speeds. As can be seen in Figure 5b, similar effective efficiency variations have also been observed for 3/4 FDR. It can be seen from this figure that the brake thermal efficiency increases at all openings and engine speeds. However, brake thermal efficiency values are lower than NDF values for 1st and 5th openings at 4000 rpm. These results are in agreement with the previous studies (Abu-Qudais et al. 2002; Zhang et al. 2011). Abu-Qudais et al. (2002) indicated that EF improved brake thermal efficiency (BTE) approximately 7.5 % over the entire speed range (from 1000 to 2000 rpm). Zhang et al. (2009) also reported 7 % reduction in BTE at low engine load but % 3 increment in BTE at high engine load for 20 % EF.



Figures 5. (a and b) Variations of bsfc and effective efficiency.



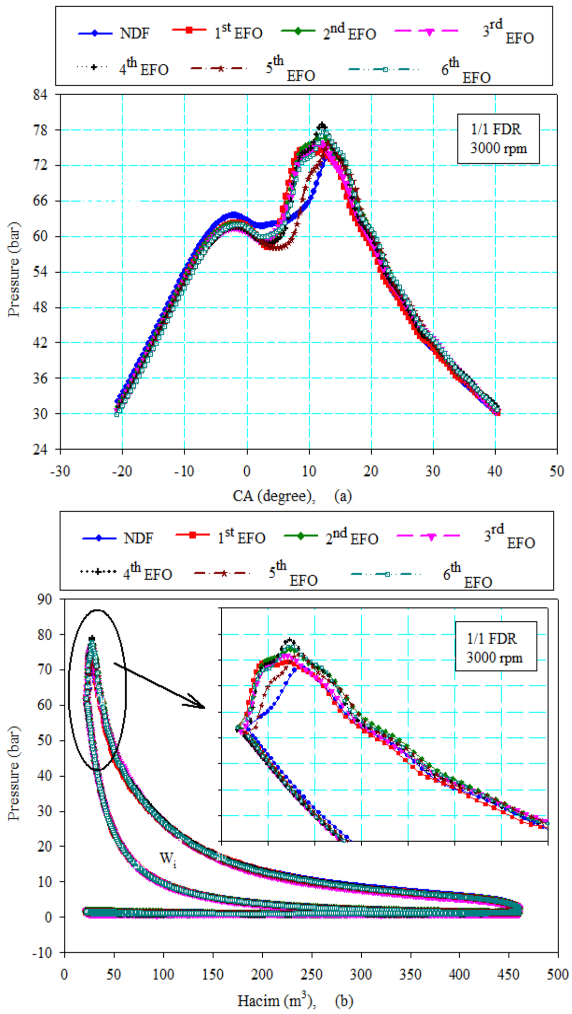
Figures 6. (a and b) Variations of the variation ratios of bsfc versus to engine speed at 1/1 and 3/4 FDR for different EFOs (EFRs).

Effects of EF on In-Cylinder Pressure

In the Figures 7 (a and b) and 8 (a and b) some examples of obtained p-θ and p-V diagrams were given for 1/1 and 3/4 FDRs respectively focusing on the part around of TDC. In both figures it can be seen that in-cylinder pressure with EF takes lower values than that of NDF during the end of the compression process for both 1/1 and 3/4 FDRs. As ethanol vaporizes during compression process, temperature values become lower than that of NDF. Thus, it can be said that lower temperatures result in lower in-cylinder pressure values.

At 1/1 FDR, for NDF the peak pressure is 73.50 bar and it occurs at crank angle of 13.07 °CA while for EF peak pressures values are 74.60, 77.09, 75.83, 78.95, 76.06, 77.67 bar and they occur at crank angles of 11.66, 12.15, 11.89, 12.08, 13.47, 12.83 °CA respectively. Maximum pressure value for 5th EFO occurs at higher crank angle than that of NDF. But for all of other EFOs maximum pressures values take place at lower crank angles than that of NDF. From this figure it can also be seen that pressure curve for EF in the high-pressure region change more sharply compared to NDF. Because ethanol has lower cetane number, higher ignition temperature and higher latent heat of vaporization than DF, EF might increase the ignition delay. Consequently, more fuel would burn in the premixed combustion mode, which could lead to higher in-cylinder pressures (Abu-Qudais et al. 2002; Zhan et al. 2009).

Moreover after start of the efficient combustion approximately near to TDC, combustible mixture could be prepared very quickly for EF because of additional gas motions produced by instantaneous burning of ethanol. Thus, as injected DF mixes with air faster and more homogenously, in-cylinder pressures for EF take higher values than that of NDF and the slopes of the pressure curves for fumigation become steeper than that of NDF. Odaka et al. (1992) mentioned about Heisy and Leitz's explanations about alcohol fumigation in their paper as follows: Heisy and Leitz explained that alcohol versus to engine speed at 3/4 FDR for different EFOs (EFRs). Fumigation caused an increase in the combustion intensity which characterized by high peak pressures and rate of pressure rise. They also explained that this was attributed to increased ignition delay resulting from the charge cooling of the vaporized alcohol and the presence of a vaporized, homogeneous alcohol fuel charge that ignites immediately as combustion starts.

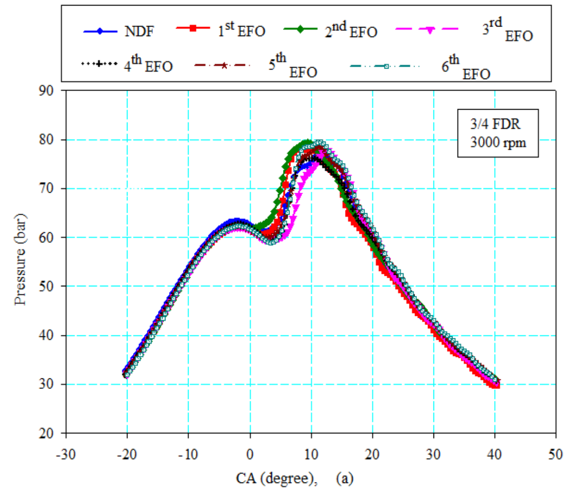


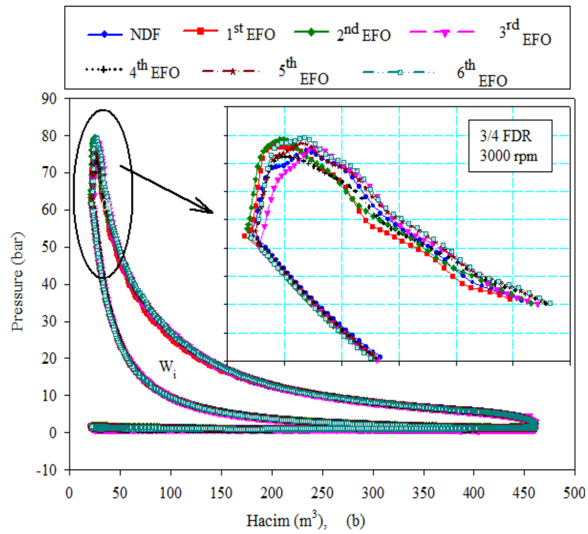
Figures 7. (a) In-cylinder pressure variations versus to crank angle at 1/1 FDR for various EFOs (EFRs). **(b)** In cylinder pressure

variations versus to cylinder volume at 1/1 FDR for various EFOs (EFRs).

In-cylinder pressure values increase somewhat for 3/4 FDR at 3000 rpm. In some case cylinder pressure values are close to NDF but they become lower than that of NDF for 3rd EFO. As shown in Figure 8a the slope of the pressure curve for fumigation near TDC is lower than that of NDF. For NDF the peak pressure is 77.19 bar and it occurs at crank angle of 11.70 °CA while for the EF the peak pressure values are 78.12, 79.29, 77.79, 76.21, 78.56 and 79.00 bar and they occur at crank angles of 8.07, 9.44, 12.38, 9.65, 11.22 and 11.53 °CA respectively.

The work delivered to the piston during compression, combustion and expansion strokes for crank angles between of 180° before TDC and 180° after TDC is known as gross indicated work or simply as indicated work. It is also defined as useful work produced in an engine cylinder (Heywood 1998; Adnan et al. 2012). Gross indicated work can be determined by calculating of enclosed area of positive part of p-V diagram. Figures 7b and 8b show these areas correspond to the variations of gross indicated works (in kJ) for various EFRs, for 1/1 and 3/4 FDRs at 3000 rpm, respectively.





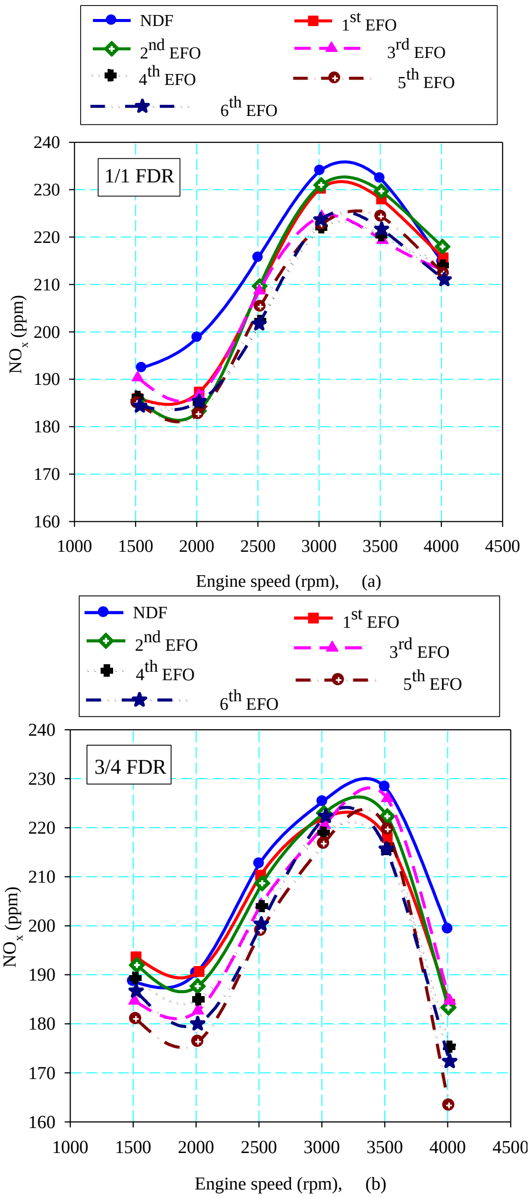
Figures 8. (a) In-cylinder pressure variations versus to crank angle at 3/4 FDR for various EFOs (EFRs). **(b)** In cylinder pressure variations versus to cylinder volume at 3/4 FDR for various EFOs (EFRs).

In the present study, to convert the pressure signals measured by using pressure sensor to pressure values (in bar), a computer code was written in Matlab by a Computer Engineer and p-θ and p-V diagrams were determined and drawn. Also, here indicated power values (in kW) was determined by computing enclosed positive areas of these p-V diagrams by using trapezoidal rule. For example, the indicated power is 43.01 kW for NDF, while the indicated power for EF become (43.55, 43.31, 43.89, 43.63, 42.40 and 44.03) kW for 1/1 FDR. These values show that indicated power increases with EF. The indicated power calculation results show that less than 1% and 2% change in power output occurred by EF at 3000 for 1/1 FDR.

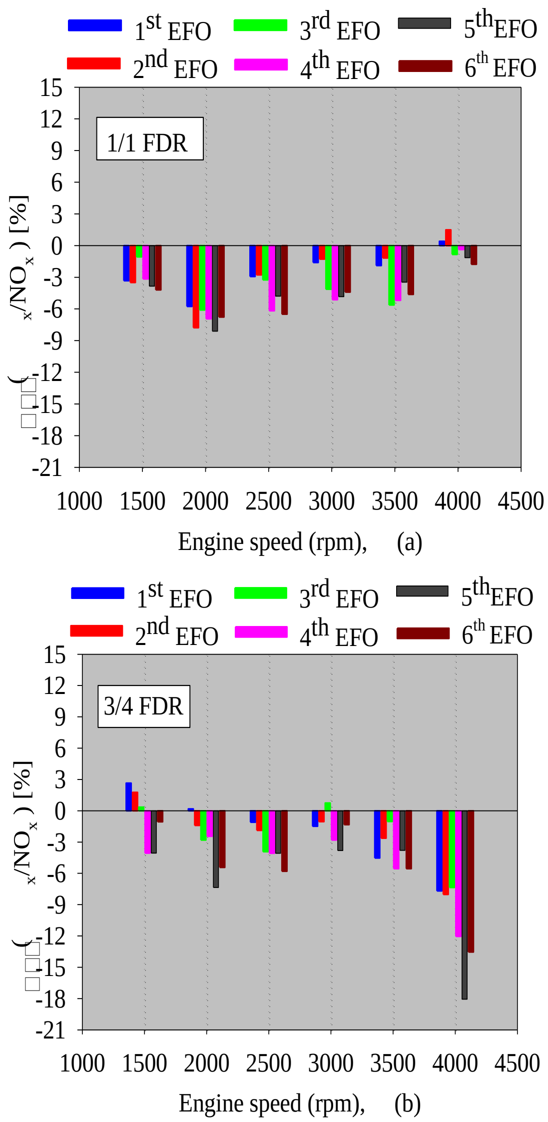
Effects of EF on NO_x Emission

Figures 9 (a and b) show the variation of NO_x emission with engine speed for six different ethanol injection conditions at 1/1 and 3/4 FDRs, respectively. Also, Figures 10 (a and b) demonstrate the variation ratios of NO_x with engine speed at six different EFOs. As can be seen from Figures 9 (a and b), NO_x emission increases as the engine speed increases, reaches its maximum values and then decreases at higher engine speeds. This can be explained on the basis that, at higher engine speeds gas motions within combustion chamber increases and this in turn leads to a faster mixing between air and fuel which results in the shortening of the ignition delay. The reduction of ignition delay minimizes the reaction time of free nitrogen and oxygen molecules in the combustion chamber which is the main mechanism of NO_x formation (Tesfa et al. 2012).

Figures 10 (a and b) clearly depict that NO_x emission takes smaller values than that of NDF at 1/1 and 3/4 FDRs for selected EF interval. For 1/1 FDR, reduction level of NO_x with EF is more significant at higher EF ratios. The reduction ratio values of NO_x emission with EF are significant at all engine speeds, except for 4000 rpm. As shown in Figure 10a, the average reduction of NO_x emission is up to 3.68 % and the maximum reduction of 8.12 % obtained at the engine speed of 2000 rpm for ~8% EF. As can be seen in Figure 10b, the similar decrement ratios of NO_x have also been obtained for 3/4 FDR. By inspection of Figure 10b; we can say that the reduction ratios of NO_x for low EF percentages are lower than that of high EF percentages. For 3/4 FDR, the average reduction of NO_x emission is up to 4.0% and the maximum decrement ratio has been obtained as 18.06 % at 4000 rpm for ~7.3% EFR. These results agreed with the observations reported in the relevant literature (Chauhan et al. 2011; Sahin and Durgun 2007; Chapman et al. 2008). Chauhan et al. (2011) investigated EF in a naturally aspirated small capacity DI diesel engine. They found that at full load NO_x emission decreases for up to 16% EF then it starts to increase by increasing of EF. Also Chapman et al. (2008) found 10% reduction in NO_x emission for dimethyl ether fumigation in a common-rail turbocharged DI diesel engine. Almost similar decrement ratios for NO_x have been given for DF and methanol fumigation in the literature (Zhang et al. 2011; Odeka et al. 2008; Sahin et al. 2008).



Figures 9. (a and b) Variations of NO_x concentration versus to engine speed at 1/1 and 3/4 FDRs for different EFOs (EFRs) respectively.

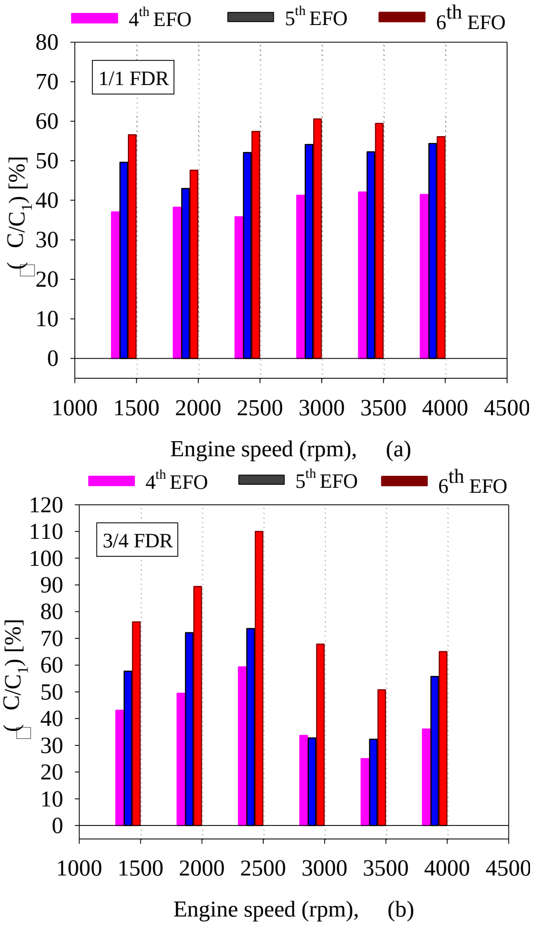


Figures 10. (a and b) Variations of the variation ratios of NO_x emission versus to engine speed at 1/1 and 3/4 FDRs for different EFOs (EFRs) respectively.

It is thought that reduction of NO_x has been arisen by improving of EF the combustion process. That is, fast burning of the air-ethanol mixture would create additional gas motions and by this effect DF injected after this event could probable mix with air more effectively and homogeneously. Thus, it is thought that, being the combustion would spread thought into combustion chamber and local peak temperature values would not be occurred, the formation of NO_x would decrease. Moreover, in various sources it was explained that, in homogeneous charge compression ignition engines (HCCI) in which a similar method to the fumigation has been applied, NO_x and soot formation decrease (Merker et al. 2006). Furthermore, long ignition delay promotes the premixed portion of the combustion process, which decreases the diffusive burning and hence also contributes to the reduction in the NO_x emission (Zhang et al. 2011; Tesfa et al. 2012).

Effects of EF on the Fuel Cost

By using equation 4, a practical cost analysis was performed in the presented study. Numerous EF and also ethanol-diesel fuel blend studies have been done in literature. But any cost analysis or cost comparison has not been found in these studies yet. Here, variation ratios of the fuel cost compared to NDF at 1/1 and 3/4 FDRs for various EFOs, especially which give best results for engine performance and NO_x emissions, were presented in the Figure 11 (a and b). As can be seen in Figure 11a that; for 1/1 FDR the fuel cost takes higher values than that of NDF at all openings and engine speeds. Actually, bsfc decreases by the improving effects of EF. But, because of being this decrement relatively small and being the price of ethanol higher than DF (in Turkey, the price of ethanol is 7.11 times of DF) total fuel cost becomes higher than DF. For 3/4 FDR, EF is more expensive at all engine speeds and at selected EFOs.



Figs.11. (a and b) Variation ratios of cost versus to engine speed at 1/1 and 3/4 FDRs for different EFOs (EFRs) respectively.

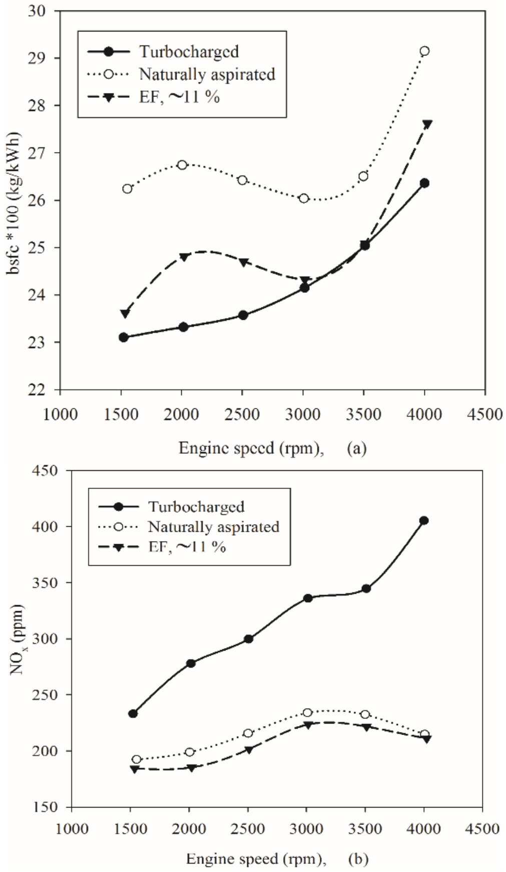
Comparison of EF and Turbocharging

It can be seen from Figures 12a-13a that, bsfc for EF takes values less than that of naturally aspirated engine but higher that turbocharged version of this engine for 1/1 and 3/4 FDRs respectively. On the other hand, as can be seen Figures 12b-13b, by EF NO_x emission take values of ~11 % for 1/1 FDR and ~10 % for 3/4 FDR and it clearly becomes lower than that of both turbocharged and naturally aspirated engine versions. Thus it can be concluded that EF improves engine performance parameters and NO_x emission but does not increase effective power as much as turbocharged version.

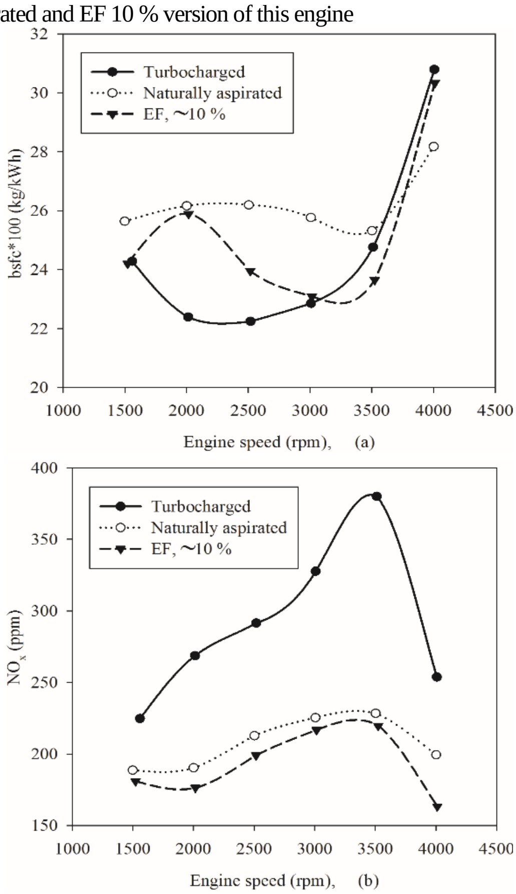
CONCLUSIONS

In the present study the effects of EF on combustion, performance and NO_x emission were investigated experimentally and compared with NDF in a high speed IDI, naturally aspirated automotive diesel engine installed at the authors' laboratory. Based on the experimental results the main effects of EF are summarized as follows:

1. Ethanol is a promising renewable oxygenated fuel for internal combustion engines and has been paid more attention in many countries. It is known from the relevant literature that when ethanol is used by fumigation method in



Figures 12. (a and b) Variations of bsfc and NO_x concentration versus to engine speed at 1/1 FDR for turbocharged, naturally aspirated and EF 10 % version of this engine



Figures 13. (a and b) Variations of bsfc and NO_x concentration versus to engine speed at 3/4 FDR for turbocharged, naturally aspirated and EF 10 % version of this engine.

diesel engine, engine performance improves and exhaust emission decreases significantly. The present study shows that EF reduces NO_x emission and bsfc and also, improves somewhat the effective power of an IDI automotive diesel engine.

2. By application of EF in this engine, NO_x emission reduces approximately at the levels of 3.68 % and 4.0 % for 1/1 and 3/4 FDRs respectively. Being NO_x one of the most important pollutants which is a poisonous compound, this is a promising results in the point of view of environmental pollution. It is known from literature that prolonged exposure to diesel engines emissions (NO_x and also soot) above a specific level should be harmful to human health.

3. For 1/1 FDR, effective power decreases at low engine speeds but it increases at high engine speeds for (1.5- 6) % ethanol percentages. However, on average 2.71 % increment in effective power is obtained for (7.74-11.16) % ethanol percentages at all of the engine speeds. bsfc decreases with ethanol percentages at all of the engine speeds and on average 5.3 % reduction in bsfc is obtained for all of the EF ratios and engine speeds. Also, effective efficiency increases with increasing EF and for example, on average 9.48 % increment ratio in effective efficiency was obtained for ~9.97% and ~11.16 % EFRs. Based on the above results, the optimum percentages of ethanol appear to be ~(7.5-11) % for this FDR. For 1/1 FDR; ~(7.5-11) % ethanol addition into the intake manifold reduces NO_x emission and improves engine performance for this engine. However, the combined fuel cost for ~(7.5-11) % ethanol addition becomes expensive.

4. For 3/4 FDR; ethanol addition into the intake manifold reduces bsfc and increases effective efficiency and on average approximately 4.4 % reduction in bsfc is obtained for all of the openings and engine speeds. However, generally EF results in somewhat decrement of effective power. However ~ (7.5-12) % ethanol addition into the intake manifold can be applied at (1500-3000) rpms engine speeds for this engine. In this operating conditions, NO_x emission and bsfc decrease simultaneously without effective power penalty.

5. EF increases in-cylinder pressure for 1/1 and 3/4 FDRs. This may be attributed to increased ignition delay resulting from the charge cooling of the vaporized ethanol and the presence of a vaporized, homogeneous ethanol-air charge which ignites immediately as combustion starts. For this reason, it was determined that the pressure variation for EF in the high-pressure region change more sharply compared to NDF.

6. Based on the above conclusions it can be decided that EF can be effectively employed in existing IDI automotive diesel engine to improve engine performance and to reduce NO_x emission. For applying this method, an adapted carburetor was used to introduce ethanol into the intake air and any other modification on the engine was not required. Thus, this method can be applied practically and economically. But, by applying EF it would not obtained effective power increment as high as turbocharging.

7. For obtained generalized results, more experimental studies must be done by using different vehicle diesel engines.

8. Authors are studying on preparing next paper which include heat release analysis and flammability analysis for EF in the diesel engine used in the presented study.

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